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Performance, Emission and Combustion Characteristics of Jatropha Oil Blends in a Direct Injection CI Engine

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ABSTRACT

Vegetable oils have energy content suitable to be used as compression ignition (CI) engine fuel. However, several operational and durability problems of using straight vegetable oils in CI engines are reported in the literature, which are primarily caused by their higher viscosity and low volatility compared to mineral diesel. The viscosity can be brought in acceptable range by (i) chemical process of transesterification, (ii) blending of oil with mineral diesel or (iii) by heating the vegetable oil using exhaust gas waste heat. Reduction of viscosity by blending or exhaust gas heating saves the chemical processing cost of transesterification.

Present experimental investigations were carried out for evaluating combustion, performance and emission behavior of Jatropha oil blends in unheated conditions in a direct injection CI engine at different load and constant engine speed (1500 rpm). Analysis of in-cylinder pressure rise, instantaneous heat release and cumulative heat release was carried out. All test blends exhibited similar combustion stages as mineral diesel; however, Jatropha oil blends showed earlier start of combustion but lower heat release rate during premixed combustion phase for all engine loads. The crank angle position of peak cylinder pressure for vegetable oil blends shifts towards top dead center compared to baseline diesel. Combustion duration was found to be comparable with diesel up to 20% concentration of Jatropha oil in the fuel. HC, CO and NO emissions were found to slightly increase with increase in Jatropha oil content in the fuel blends.

INTRODUCTION

Diesel engines are the most efficient engines commonly available today. Diesel engines have provided power units for transportation systems (passenger cars, buses etc.), goods transportation systems (trucks etc.), ships, railway locomotives, non-road equipment used for farming and construction, and in almost every type of industry due to its economy of operation and durability. They move a large portion of the world's goods, power much of the world's equipment, and generate electricity more economically than any other device in their size range. Energy insecurity caused by depleting petroleum resources and environmental issues of fossil fuels have generated urgency for the alternative renewable compression ignition engine fuels. However diesel engines are one of the largest contributors to environmental pollution problems worldwide.

Alternative fuels should be easily available, environment friendly and techno-economically competitive. Successful alternative fuel should fulfill environmental and energy security needs without sacrificing engine operating performance [1]. Renewable resources offer the opportunity to tap local and renewable resources and reduce dependence on imported energy resources. For the developing countries of the world, fuels of bio-origin provide a feasible solution to the twin crises of fossil fuel depletion and environmental degradation. The idea of using vegetable oils as fuel for diesel engine is not new. When Rudolf diesel invented diesel engine, he demonstrated it at the 1900 world exhibition in Paris, employing peanut oil and said "The use of vegetable oils for engine fuels may seem insignificant today, but such

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oils may become in course of time as important as petroleum and coal tar products of present times” [2]. With the advent of cheap petroleum, appropriate crude oil fractions were refined to serve as fuel thus diesel fuels and diesel engines started evolving together.

Vegetable oils mainly contain triglycerides (90~98%) and traces of mono and diglycerides. Triglycerides consist of three fatty acid molecules and a glycerol molecule. They contain significant amounts of oxygen [3-4]. The fatty acids vary in their carbon chain length and number of double bonds present in their molecular structure. Vegetable oils contain free fatty acids (generally 1–5%), phospholipids, phosphatides, arotenes, tocopherols, sulfur compounds and traces of moisture. Commonly found fatty acids in vegetable oils are stearic, palmitic, oleic, linoleic and linolenic acid. Vegetable oils can be produced even on a small scale for on-farm utilization to run tractors, pumps and small engines for power generation/ irrigation. Suitability of vegetable oils as fuels for diesel engines depends on their physical, chemical and combustion characteristics as well as the type of engine used and operating conditions [4].

Vegetable oils can be used directly or blended with diesel to operate compression ignition engines. Use of blends of vegetable oils with diesel has been used successfully by various researchers in several countries [5-11]. It has been reported that use of 100% vegetable oil is also possible with minor fuel system modifications [12]. Short-term engine performance tests have indicated good potential for most vegetable oils as fuel. The use of vegetable oil results in increased volumetric fuel consumption and BSFC. Emissions of CO, and HC were found to be higher, whereas NOx and particulate emissions were lower compared to mineral diesel [13-18]. In waste cooking oil study SOx emissions from vegetable oils were higher than that from mineral diesel fuelled engine [15].

Undoubtedly, transesterification is well accepted and best suited method of utilizing vegetable oils in CI engine without significant long-term operational and durability issues. However, this adds extra cost of processing because transesterification reaction involves expensive chemical and process heat as inputs. In rural and remote areas of developing countries, where grid power is not available, vegetable oils can play a vital role in decentralized power generation for irrigation and electrification. In these remote areas, different types of vegetable oils are locally produced but it may not be possible to chemically process them due to logistics issues in rural settings. Hence using heated or blended vegetable oils as petroleum fuel substitutes is an attractive proposition. Keeping this in mind, a set of engine experiments were conducted using Jatropha oil in an engine, which is typically used for agriculture, irrigation and decentralized electricity generation worldwide.

Blending as a technique was used to lower the viscosity of Jatropha oil in order to eliminate various operational difficulties. The present research is aimed at exploring technical feasibility of Jatropha oil in direct injection compression ignition engine without any substantial hardware modifications.

EXPERIMENTAL SETUP

This is an example of a Main Heading section. This section will include sub-sections. Four-stroke, single cylinder, constant-speed, water-cooled, direct injection CI engines (Make: Kirloskar Oil Engines Ltd. India; Model: DM-10) was used to study the effects of Jatropha oil blends on performance, emissions and combustion characteristics (Table1). The engine was operated at a constant speed of 1500 rpm. The inlet valve opens 4.5° before TDC and closes 35.5° after BDC. The exhaust valve opens 35.5° before BDC and closes 4.5° after TDC. Fresh lubricating oil was filled in oil sump before beginning the experiment. This engine consists of gravity-fed fuelling system with efficient paper element filter, force-feed lubrication for main bearing, large-end bearings and camshaft bush, and run-through/ thermo-siphon cooling system.

A piezoelectric pressure transducer (Make: Kistler Instruments, Switzerland; Model: 6613CQ09-01) was installed in the engine cylinder head to acquire the combustion pressure–crank angle history. Machining for installation of pressure transducer was done in cylinder head and the engine main shaft was coupled with a precision shaft encoder (Make: Encoder India Limited, Faridabad, Model: ENC58/6-720ABZ/5-24V). Signals from the pressure transducer were amplified using a charge meter (Make: Kistler Instruments, Switzerland; Model: 5015A). The high-precision shaft encoder was used for delivering signals of crank angle with a resolution of 0.5° crank angle. A TDC marker was used to locate the TDC position in every cycle of the engine. The signals from the charge amplifier, TDC marker and shaft encoder were acquired using a high-speed data acquisition system (Make: Hi-Techniques, USA; Model: meDAQ). Engine tests were carried out at 1500±3 rpm, for 200 bar fuel injector pressure for diesel (D), and Jatropha oil blends (% v/v) with diesel namely 5%, 20%, 30%, 50%, 75%, 100% (J05, J20, J30, J50, J75, J100). Six engine load conditions where the combustion data were acquired were 0%, 20%, 40%, 60%, 80% and 100% (45 N-m) of rated load. The cylinder pressure data were acquired for 50 consecutive cycles and then averaged in order to eliminate the effect of cycle-to-cycle variations. All tests were carried out after thermal stabilization of the engine.

Exhaust gas opacity was measured using smoke opacimeter (Make: AVL Austria, Model: 437). The exhaust gas composition was measured using exhaust gas analyzer (Make: AVL India, Model: DIGAS 444). It measures CO_2, CO, HC, NO and O_2 concentrations in the exhaust gas.

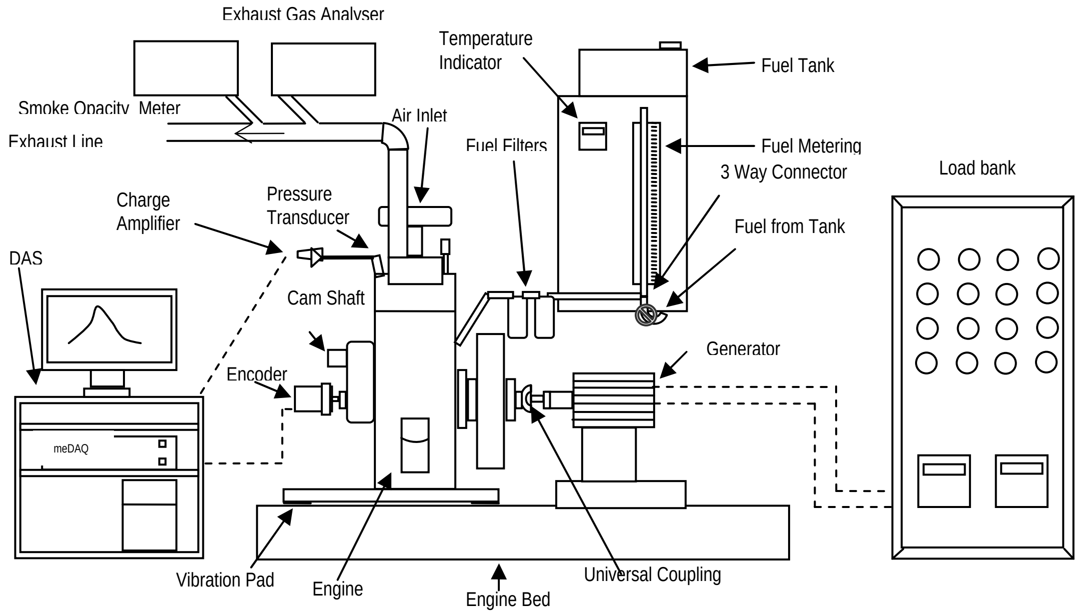

Figure 1: Schematic of experimental setup

Table 1: Engine Specifications

Engine Parameters	Specifications
Manufacturer	Kirloskar Oil Engines Ltd, India
Engine Type	Vertical, 4-stroke, single cylinder, constant speed, direct injection, CI engine
Rated power	7.4 kW at 1500 rpm
Bore / stroke	102 mm / 116 mm
Displacement vol.	0.948 liters
Compression ratio	17.5: 1
Start of Injection timing	26° BTDC
Nozzle opening pressure	200-205 bars
Cooling type	Water cooling
Length/ width/ height	685/ 532/ 850 mm
BMEP at 1500 rpm	6.34 bar (Max.)
Lubricating oil sump capacity	3.7 Liters
Injection system	Reciprocating jerk pump

Table 2: Important Properties of Diesel and Jatropha Oil

Property	Diesel	Jatropha
Density (kg/m^3)	833.7	921.8
Kinematic Viscosity @ 40^{o}C (cSt)	2.71	34.33
Calorific Value (MJ/kg)	43.06	41.85
Flash Point (^{o}C)	48	180
Carbon Residue %, (w/w)	0.08	0.74
Ash Content %, (w/w)	0.014	0.036
Carbon %, (w/w)	83.12	76.56
Hydrogen %, (w/w)	14.72	13.19
Nitrogen %, (w/w)	0.45	0.34
Copper Corrosion Grade	1a	1a

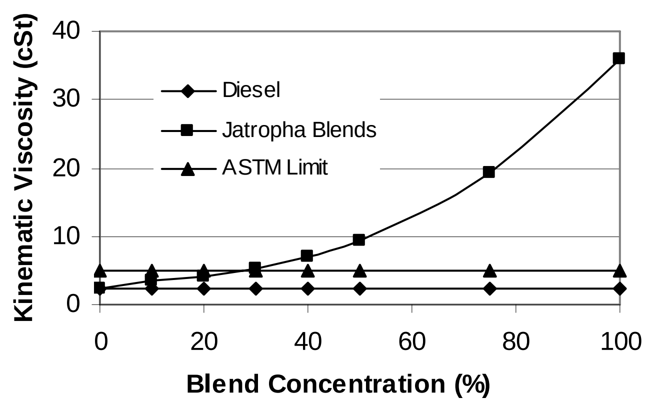

Figure 2: Viscosity of various Jatropha oil blends at 40^{o}C

RESULTS AND DISCUSSION

Important properties of Jatropha oil used in the study are compared with mineral diesel in Table 2. Figure 2 shows viscosity of Jatropha oil blends. Viscosity of blends up to 30% Jatropha oil is slightly higher than diesel but within the ASTM limits for CI engine fuels. All the performance, combustion and emission tests were carried out at optimum fuel injection pressure (200 bars) for bsfc, thermal efficiency and smoke opacity [5].

PERFORMANCE AND EMISSIONS TEST

Experiments were conducted using various blends of Jatropha oil with diesel. BSFC was found to increase with higher proportion of Jatropha oil in the blend compared to diesel in the entire load range (Figure 3).

Calorific value of Jatropha oil is lower compared to that of diesel, therefore increasing proportion of Jatropha oil in blend decreases the calorific value of the blend, which results in increased BSFC. Thermal efficiency of Jatropha blends was lower than that with mineral diesel (Figure 4). However, thermal efficiency of blends up to J20 was very close to diesel. Oxygen present in the fuel molecules improves the combustion characteristics but higher viscosity and poor volatility of vegetable oils lead to their poor atomization and combustion characteristics. Therefore, thermal efficiency was found to be lower for higher blend concentrations compared to that of mineral diesel.

Lowest CO_2 emissions were observed for diesel (Figure 5). CO_2 emissions for lower blend concentrations were close to diesel. But for higher blend concentrations, CO_2 emissions increased significantly. The emissions of total CO increases with increasing load but brake specific CO emission first decreases then increases (Figure 6). Higher the load, richer fuel–air mixture is burned, and thus more CO is produced due to lack of oxygen. For lower blends (upto J30), CO emissions for Jatropha oil are close to mineral diesel but higher blends exhibit significant increase in brake specific CO emissions. Jatropha oil blends exhibit higher HC emissions compared to diesel (Figure 7). It can be observed that HC emissions increase with increasing proportion of Jatropha oil in the blends.

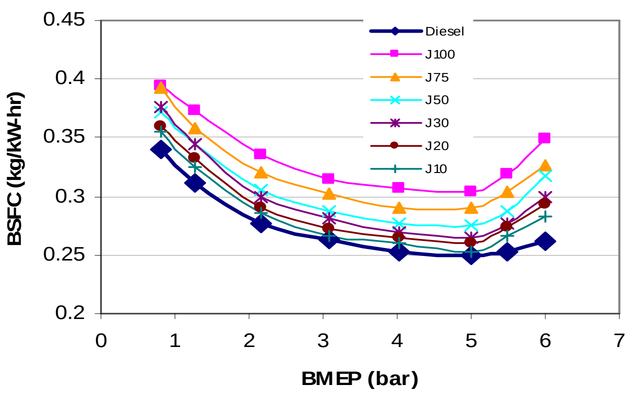

Figure 3: Comparison of bsfc of Jatropha oil blend fuelled engines

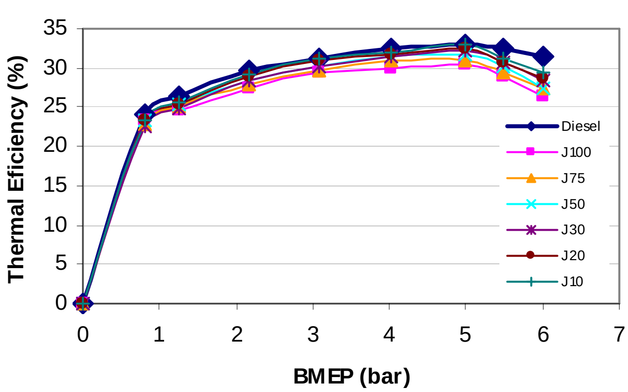

Figure 4: Comparison of thermal Efficiency of Jatropha oil blend fuelled engines

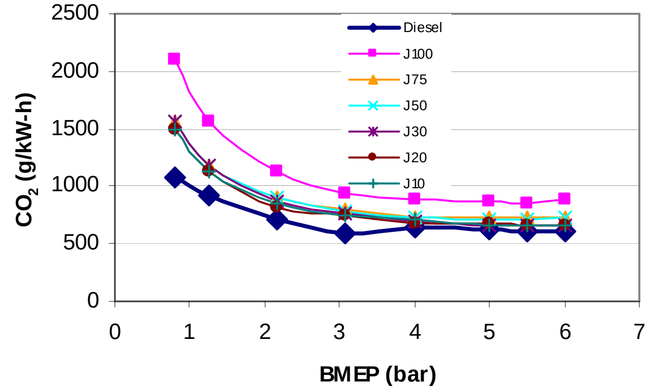

Figure 5: Comparison of CO_2 emissions of Jatropha oil blend fuelled engines

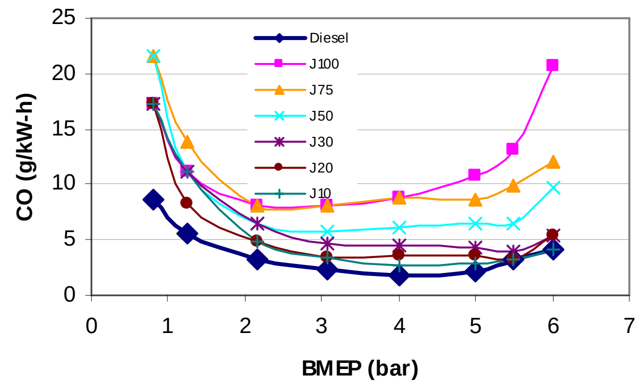

Figure 6: Comparison of CO emissions of Jatropha oil blend fuelled engines

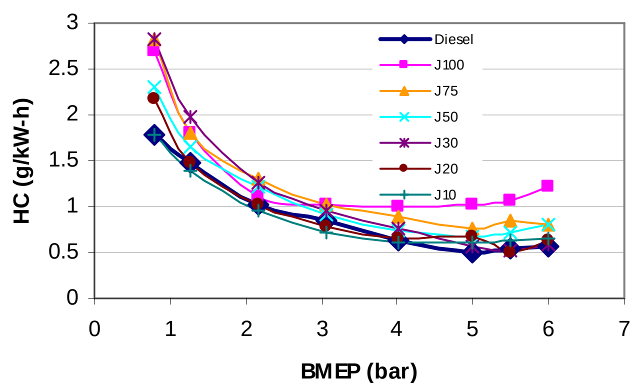

Figure 7: Comparison of HC emissions of Jatropha oil blend fuelled engines

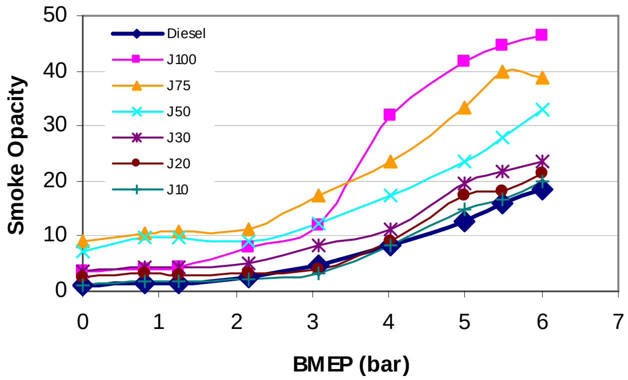

Figure 8: Comparison of smoke opacity of Jatropha oil blend fuelled engines

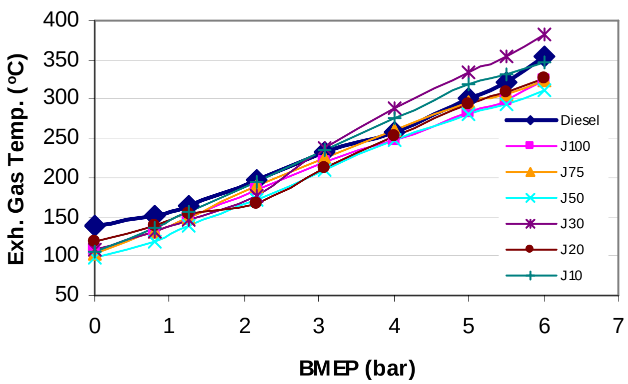

Figure 9: Comparison of exhaust gas temperature of Jatropha oil blend fuelled engines

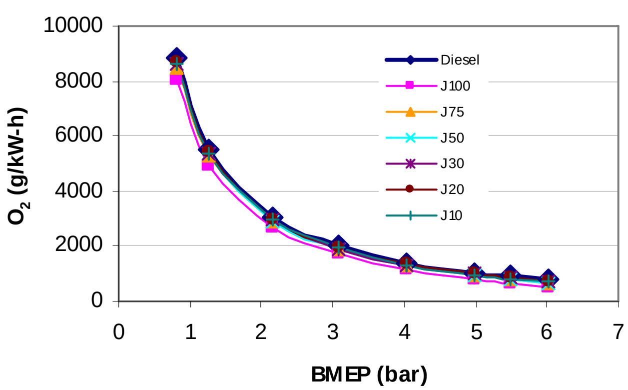

Figure 10: Comparison of oxygen content in exhaust gas of Jatropha oil blend fuelled engines

The smoke opacity increases with increase in Jatropha oil concentration in blends particularly at higher loads (figure 8). Higher smoke opacity may be due to poor atomization of the Jatropha oil. Bulky fuel molecules and higher viscosity of Jatropha oil result in poor atomization of fuel blends. The exhaust gas temperature with blends having higher percentage of Jatropha oil was higher compared to that of diesel at higher loads (figure 9). Concentration of oxygen in the exhaust gas decreases

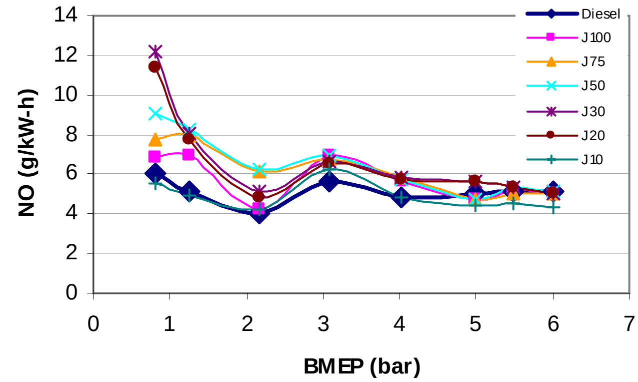

Figure 11: Comparison of NO emissions of Jatropha oil blend fuelled engines

with increase in concentration of Jatropha oil in the blends. High exhaust gas temperature and low pure oxygen in the exhaust indicate possibility of high NO formation with the increase of Jatropha oil concentration in the blends. NO was found to be minimum for J10 and NO emissions were comparable with diesel for J20 and J30. This observation may be due to slow heat release rate in case of Jatropha blends (Figure 13).

COMBUSTION ANALYSIS

(a)In cylinder pressure vs. crank angle diagram

The variations in cylinder pressure with crank angle for mineral diesel, 5%, 20%, 50% and 100% Jatropha oil at different engine operating conditions are shown in Figures 12a-f. This data is analyzed by Revelation (Hi-Techniques) combustion analysis software for results presented in figures 13-18. From these figures, it can be noticed that at low engine loads, cylinder pressure trends are almost similar for different fuel blends. Jatropha oil blends are showing relatively earlier pressure rise with respect to mineral diesel for higher engine loads suggesting lower ignition delay for Jatropha oil blends. For low loads, blends show higher peak pressure but at high engine loads, diesel gives higher peak pressure. For 80 and 100 percent rated load, 5% Jatropha blend shows slightly higher peak pressure compared to mineral diesel. At all engine loads, combustion starts earlier for Jatropha oil blends than for mineral diesel and the rate of pressure rise is slower for vegetable oil blends because of slower burning characteristics. As the engine load is increased, the start of combustion point shifts earlier for all fuels. Ignition delay represents the time taken in physical and chemical pre-flame reactions. In this study, ignition delay was not measured; however, the start of combustion may reflect the variation in ignition delay because fuel pump and injector settings were kept identical for all fuel samples.

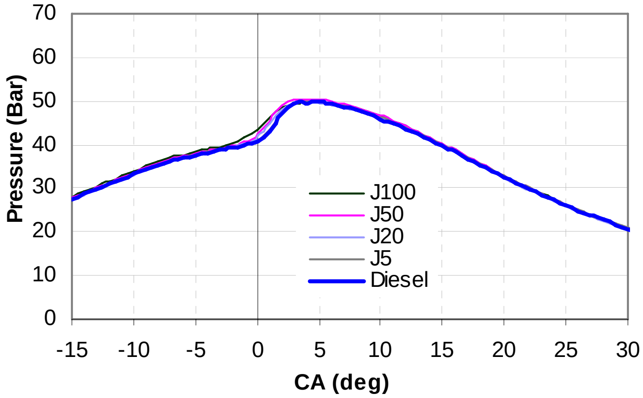

Figure 12a: Pressure–crank angle diagram for 0% rated engine Load

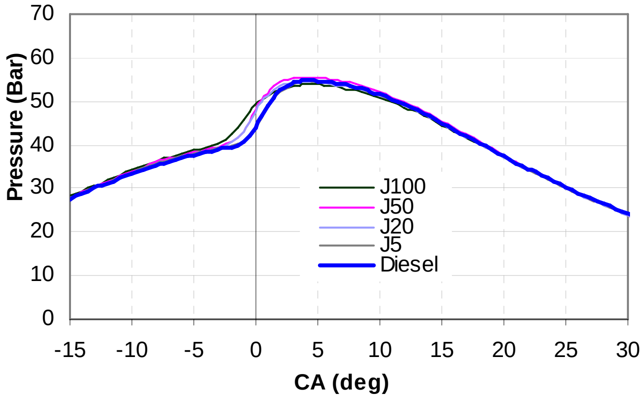

Figure 12b: Pressure–crank angle diagram for 20% rated engine Load

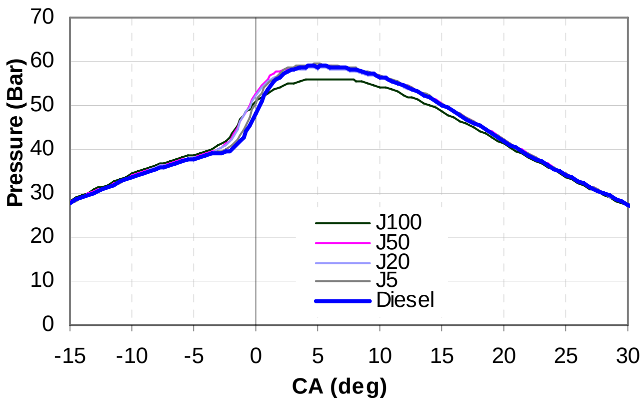

Figure 12c: Pressure–crank angle diagram for 40% rated engine Load

Combustion starts earlier for Jatropha oil (Figures 12a-f) partially owing to a shorter ignition delay and partially owing to advanced injection timing (because of a higher bulk modulus and higher density of Jatropha oil). In spite of the higher viscosity and lower volatility of the Jatropha oil, the ignition delay seems to be lower than mineral diesel. This may possibly be because a complex and rapid pre-flame chemical reaction takes place at high temperatures. As a result of the high in-cylinder temperature existing during fuel injection, Jatropha oil may undergo thermal cracking; as a result of this, lighter

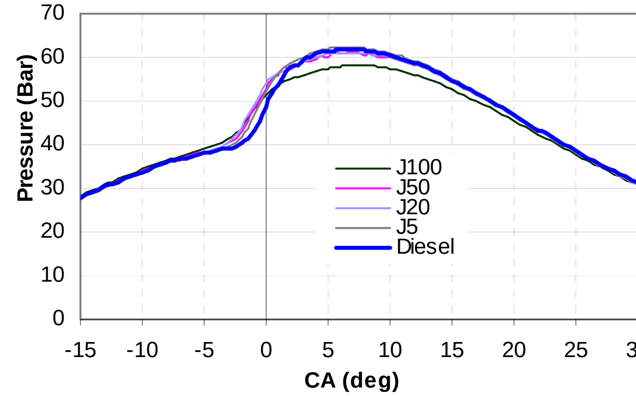

Figure 12d: Pressure–crank angle diagram for 60% rated engine Load

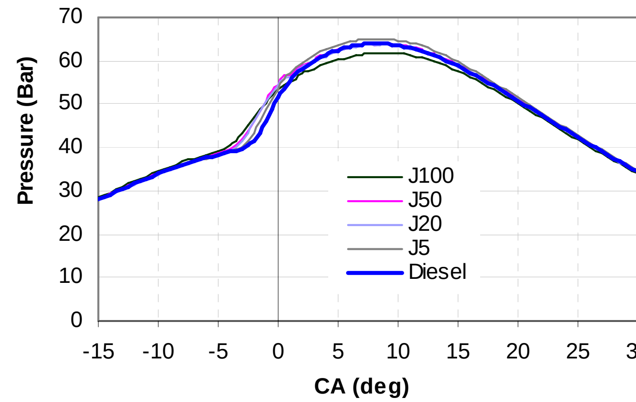

Figure 12e: Pressure–crank angle diagram for 80% rated engine Load

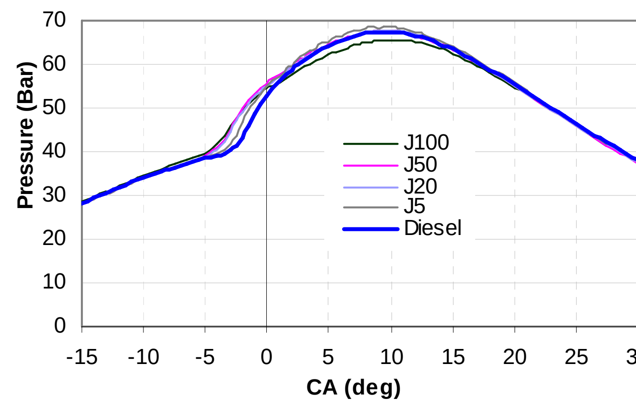

Figure 12f: Pressure–crank angle diagram for 100% rated engine Load

compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay. The combustion of Jatropha blends however seems to be slower essentially because of bulkier and complex fuel molecules of vegetable oils, which essentially take longer time for releasing the heat therefore leading to slower rate of heat release. This can be confirmed by the ROHR diagrams in following sub-section.

(b) Instantaneous rate of heat release

Figures 13a-f show the heat release rate diagrams for all Jatropha blends at different engine operating conditions. Because of the vaporization of the fuel accumulated during ignition delay at the beginning, a negative heat release is observed and, after initiation of combustion, heat release becomes positive. All Jatropha blends experience identical combustion stages as mineral diesel (such as ignition delay, premixed combustion, mixing controlled combustion or diffusion combustion, late combustion). After the ignition delay, premixed fuel air mixture burns rapidly, followed by diffusion combustion, where the burn rate is controlled by fuel–air mixing.

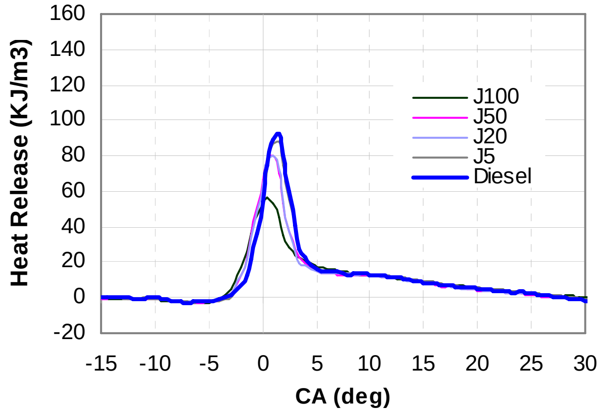

Figure 13a: Instantaneous Heat release vs. crank angle diagram for 0% rated engine Load

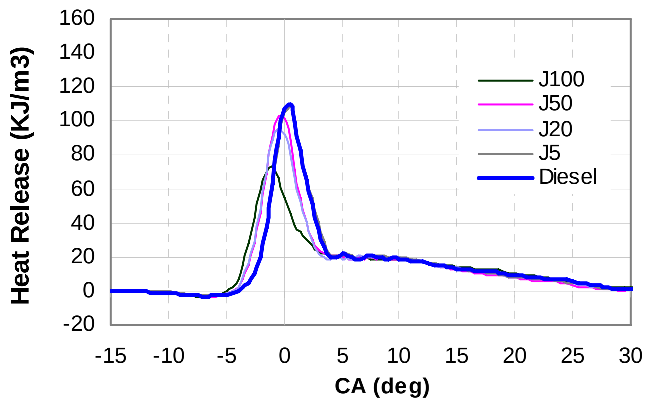

Figure 13b: Instantaneous Heat release vs. crank angle diagram for 20% rated engine Load

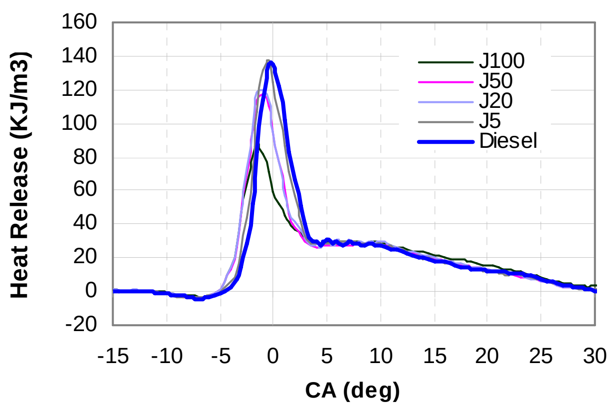

Figure 13c: Instantaneous Heat release vs. crank angle diagram for 40% rated engine Load

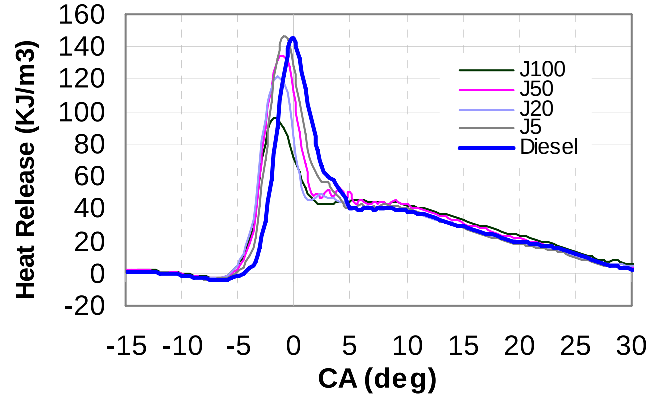

Figure 13d: Instantaneous Heat release vs. crank angle diagram for 60% rated engine Load

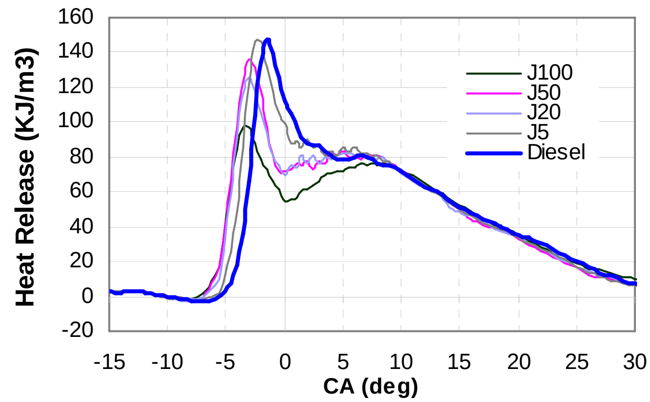

Figure 13e: Instantaneous Heat release vs. crank angle diagram for 80% rated engine Load

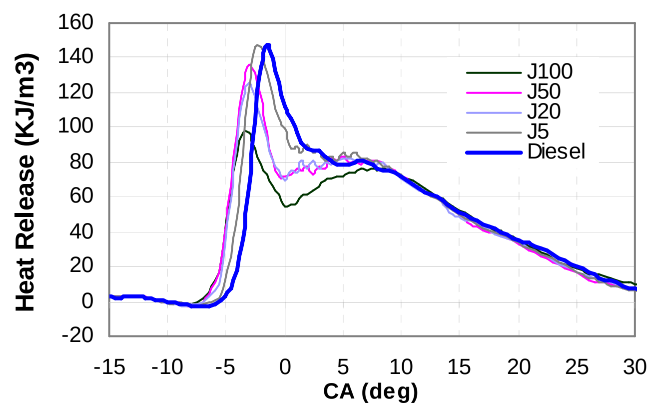

Figure 13f: Instantaneous Heat release vs. crank angle diagram for 100% rated engine Load

It can be observed that combustion starts earlier for Jatropha blends under all engine operating conditions. The premixed combustion heat release is always higher for mineral diesel owing to higher volatility and better mixing of diesel with air. Another reason may be longer ignition delay of mineral diesel, which leads to a larger amount of fuel accumulation in the combustion chamber at the time of the premixed combustion stage, leading to a higher rate of heat release. One can notice that at higher engine loads, the mixing controlled combustion is dominant (as observed by a second peak) for Jatropha oil blends. This is possibly due to longer combustion duration of larger fuel molecules of the vegetable oils.

(c) Cumulative heat release diagram

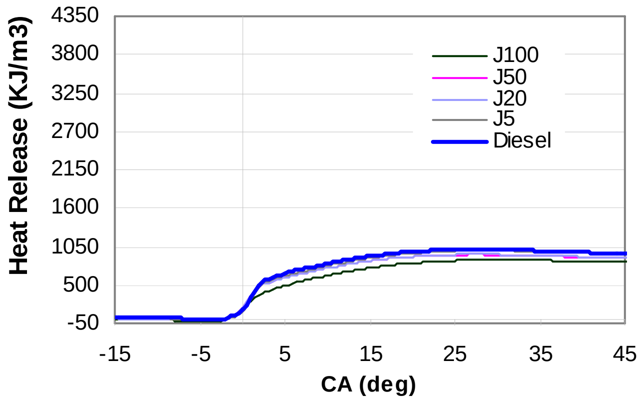

Figure 14a: Cumulative heat release vs. crank angle diagram for 0% rated engine Load

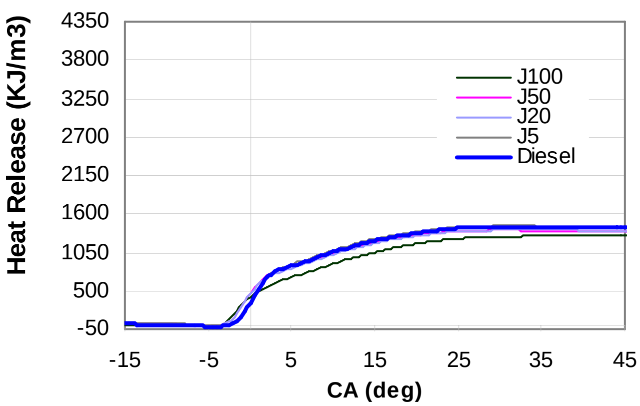

Figure 14b: Cumulative heat release vs. crank angle diagram for 20% rated engine Load

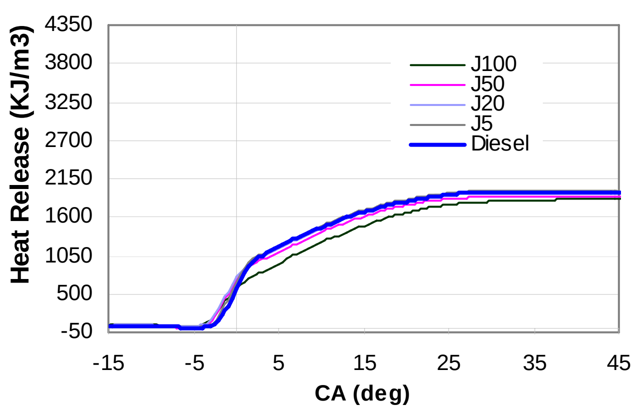

Figure 14c: Cumulative heat release vs. crank angle diagram for 40% rated engine Load

Figure 14a-f shows the cumulative heat release for different blends of Jatropha oil at different engine load conditions. These graphs show the tendency of earlier release of fuel energy for Jatropha blends, which becomes more prominent at higher engine loads. Combustion for mineral diesel starts later but quickly it exceeds the cumulative heat released for Jatropha oil blends, suggesting a faster burning of mineral diesel. Cumulative heat release decreases as the proportion of Jatropha oil increases in the blend.

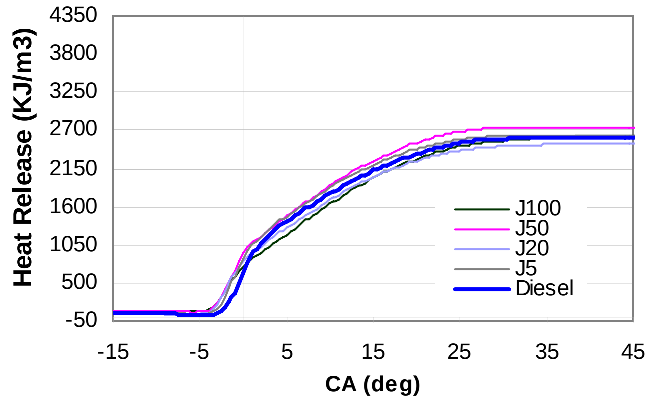

Figure 14d: Cumulative heat release vs. crank angle diagram for 60% rated engine Load

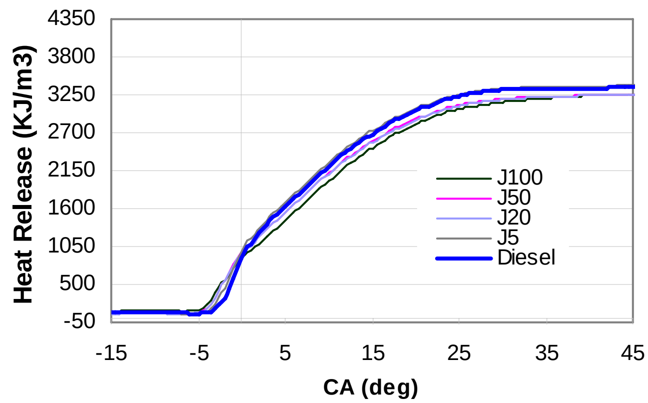

Figure 14e: Cumulative heat release vs. crank angle diagram for 80% rated engine Load

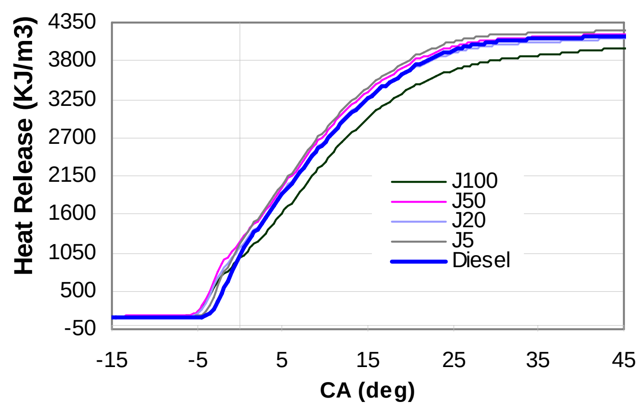

Figure 14f: Cumulative heat release vs. crank angle diagram for 100% rated engine Load

(d) Mass fraction burn crank angle

Figure 15a shows the crank angle for 5 per cent mass fraction burned. This figure shows that 5% fuel burns earlier for Jatropha oil blends and it burns successively earlier for an increasing proportion of Jatropha oil in blends. This is due to the earlier start in combustion for Jatropha oil blends, as suggested earlier i.e. progressively lower ignition delay for increasing blends of Jatropha oil compared to mineral diesel.

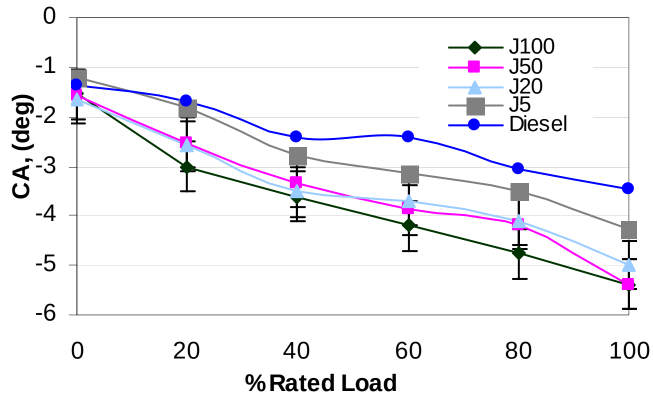

Figure 15a: Crank angle for 5% mass faction burn

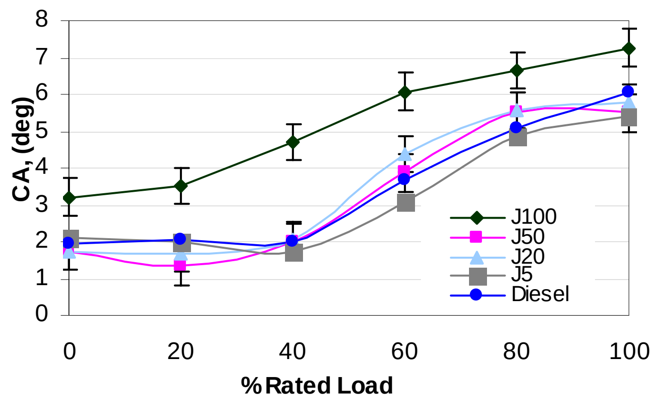

Figure 15b: Crank angle for 50% mass faction burn

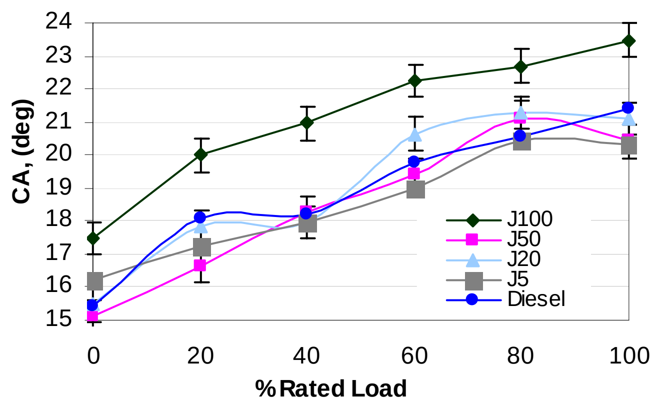

Figure 15c: Crank angle for 90% mass faction burn

As the engine load is increased, this deviation increases because, at higher loads, the combustion start crank angle decreases (Figures 12a-f). Figure 15b shows the crank angle degree for 50 percent mass fraction burned at different engine load conditions. This remains almost same for all blends and diesel except J100, where slower burning of Jatropha can be felt. This may be because Jatropha oil's flash point is higher and higher viscosity of Jatropha oil hinders atomization and vaporization of fuel. Figure 15c shows the crank angle degree for 90 percent mass fraction burned at different engine load conditions. 90 percent mass fraction burned time is comparable for Jatropha blends except J100 because blends contain mineral diesel, which possibly accelerate the combustion process. More fuel mass is required in case of Jatropha oil blends because of lower calorific value of these blends vis-à-vis mineral diesel. These factors lead to longer combustion duration for Jatropha oil blends compared to mineral diesel.

(e) Crank angle for maximum pressure and maximum pressure rise rate

Figure 16a shows the maximum cylinder pressure at different loads for different blends. It shows that at higher engine loads the peak pressure for mineral diesel is comparable to Jatropha oil blends except J100. In case of different Jatropha oil blends, the difference in peak pressure is not significant. For Jatropha oil blends, mineral diesel's combustion delay and volatility of Jatropha oil causes comparable peak pressure. The location of this peak pressure (Figure 16b) is also comparable for all Jatropha oil blends with that of mineral diesel and this is within a narrow band of 3 crank angle degrees of all the blends under investigation. Maximum cylinder pressure is attained within 1–10 crank angle degree after the TDC for all blends under different load conditions (figure 16b). At very low engine loads (particularly idling and 20 per cent rated load) because of the longer ignition delay, combustion starts later for mineral diesel than Jatropha oil blends. As evident from the pressure crank angle diagram at no-load condition for mineral diesel (Figures 12a-b), combustion starts near TDC at 1500 rpm. As a result, the peak cylinder pressure attains a lower value as it is further away from the TDC in the expansion stroke at low engine loads (Figure 16b).

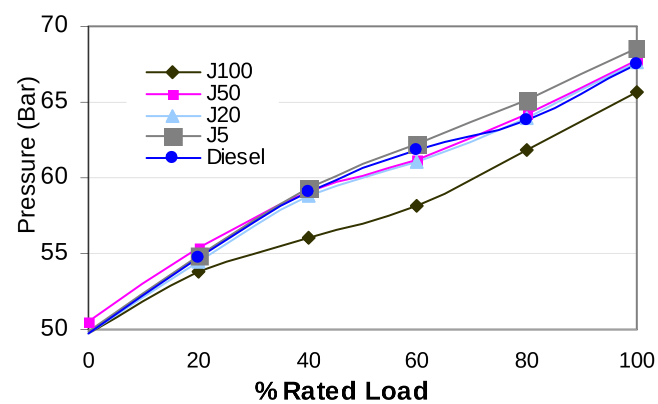

Figure 16a: Maximum in-cylinder pressure for different engine loads

Figure 17a shows the variation in the rate of pressure rise ($dP/d\theta$) with engine loads for all fuels. The rate of pressure rise varies from 4 bars/ deg at lower engine loads to 6 bars/deg at higher engine loads. Rate of pressure rise decreases as the fraction of Jatropha oil increases in the blend. This is because Jatropha oil contains heavier hydrocarbon molecules which have a higher boiling range and lower volatility. At no load, the rate of pressure rise for diesel is slightly lower than Jatropha oil blends because at this engine condition, a very small quantity of fuel is injected into the combustion chamber and combustion starts after TDC (Figure 12a) for mineral diesel, having a slightly higher delay period as mentioned earlier.

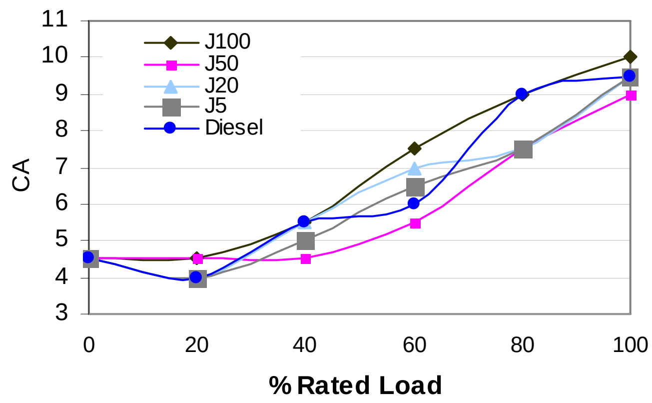

Figure 16b: Crank Angle at maximum in-cylinder pressure for different engine loads

However, the rate of pressure rise is higher for mineral diesel at higher engine loads (Figure 17a) because of higher rate of heat released during premixed combustion.

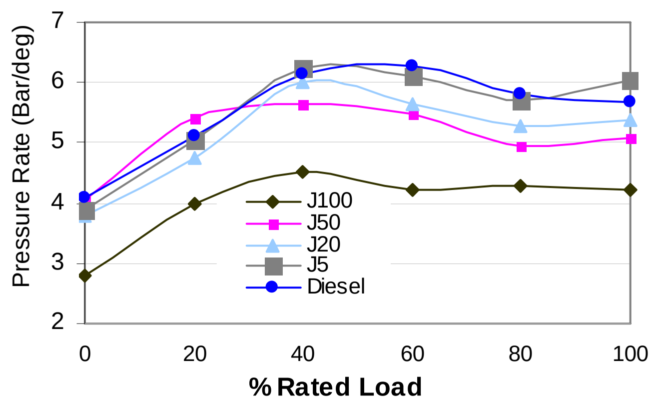

Figure 17a: Maximum rate of pressure rise for different engine loads

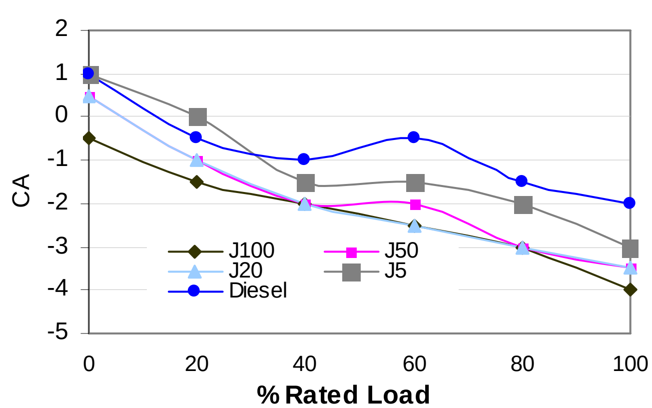

Figure 17b: Crank angle for maximum rate of pressure rise for different engine loads

Figure 17b shows the crank angle, at which the peak cylinder pressure is attained for all fuels at different engine operating conditions. It can be observed that the rate of heat release is progressively earlier for Jatropha blends and also its magnitude is lower compared to mineral diesel. The maximum rate of pressure rise for Jatropha oil and its blends are lower than mineral diesel. Maximum rate of pressure rise is slightly higher for J50 for lower loads because of lower ignition delay. Slightly higher maximum rate of pressure rise for J5 at high load is due to presence of volatile material that with high flash point leads to high rate, but overall satisfactory operation of compression-ignition engine with these blends.

(f) Combustion duration

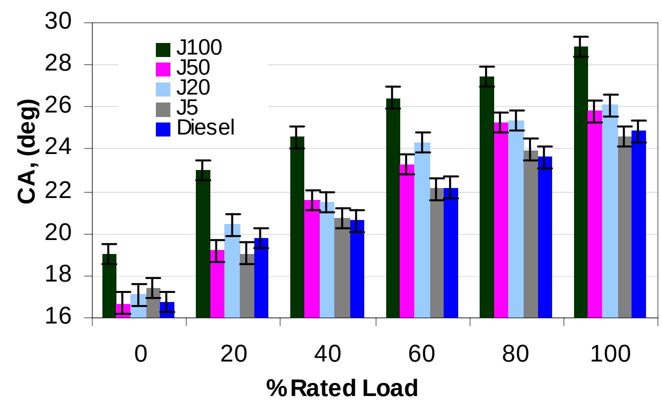

Figure 18: Combustion Duration for unheated Jatropha oil and blends vis-à-vis mineral diesel

Figure 18 shows the variation in combustion duration for different blends at different engine loads. Crank angle duration from 5 per cent mass burn to 90 per cent mass burn has been taken as the combustion duration for comparing different fuels. Combustion duration increases with increases in the engine load owing to the increase in the quantity of fuel injected. Combustion duration was observed to be higher for Jatropha oil blends than for mineral diesel. There is increase in combustion duration with the increase in the proportion of Jatropha oil in the blends again reaffirming slower combustion characteristics of Jatropha oil.

CONCLUSIONS

The performance and emissions tests were conducted with mineral diesel and blends of Jatropha oil at different loads at constant speed (1500 rpm). From the experimental results obtained, Jatropha blends of 20% (v/v) or less are found to be promising alternative fuels for compression ignition engines. These blends can be directly used as straight vegetable oil as a partial replacement of mineral diesel and do not require any major modification in the engine hardware. BSFC and exhaust gas temperatures for Jatropha oil blends were found to be higher compared to mineral diesel. Thermal efficiency was slightly lower for Jatropha oil blends compared to diesel. CO_2, CO, HC, and smoke opacity were marginally higher for lower Jatropha oil blends (upto J20) compared to that of mineral diesel. Emission parameters such as smoke opacity, CO_2, CO, and HC were found to have increased with increasing proportion of Jatropha oil in the blends compared to diesel. NOx emissions were minimum for 10% Jatropha blend.

Direct-injection stationary diesel engine was operated under steady state, at different engine loads at 1500 rpm to investigate the combustion characteristics of Jatropha oil blends vis-à-vis diesel. Experiment show that the combustion phases are almost similar for Jatropha oil blends (lower) and mineral diesel. J100 shows lower combustion delay however slower heat release rate. Combustion duration for Jatropha oil blends is higher than mineral diesel and it increases as engine load increases. In-cylinder pressure was observed to be higher for mineral diesel under all load conditions, but J5 shows slightly higher peak pressure than mineral diesel whereas other Jatropha blends were on lower side. Detailed combustion analysis suggests that J5 to J20 gives exactly identical combustion as that of mineral diesel in the unmodified engine to partially replace mineral diesel without engine hardware modification.

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